A

Project Report

On

Implementation of Traditional and Non-Traditional Optimization Algorithms for Heat Exchanger Design

Submitted by

Gaurav Singh

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Under the guidance of

Dr. Madhusree Kundu

DEPARTMENT OF CHEMICAL ENGINEERING

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CERTIFICATE

*This is to certify that the project report entitled, "*Implementation of Traditional and Non-Traditional Optimization Algorithms for Heat Exchanger Design*", submitted by* ***Gaurav Singh****(109CH0492) in partial fulfillments for the requirements for an award of Bachelor of Technology Degree in Chemical Engineering at National Institute of Technology, Rourkela is prepared by him under my supervision and guidance and this work has not been submitted elsewhere for a degree.*

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Place:

Dr. Madhusree Kundu

(Thesis Supervisor)

Dept. of Chemical Engg.

NIT Rourkela

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Gaurav Singh

ABSTRACT

The transfer of heat to and from process fluids is an essential part of most of the chemical processes. So the Heat Exchangers (HEs) are used extensively and regularly in the process and allied industries and are very important during design and operation. The most commonly used type of heat exchangers are double pipe heat exchangers and shell-and-tube heat exchangers. Shell-and-tube heat exchangers are used extensively in engineering applications like power generations, refrigeration and air-conditioning, petrochemical industries etc. These heat exchangers can be designed for almost any capacity. A primary objective in the heat exchanger (HE) design is the estimation of the minimum heat transfer area required for a given heat duty, as it governs the overall cost of the heat exchanger. However, many number of combinations of the design variables are possible. The design variables in a double pipe heat exchanger are-inner pipe diameter and thickness, outer pipe diameter and length of the exchanger. The design variables in a shell and tube heat exchanger are- tube outer diameter, tube pitch, tube length, number of tube passes, baffle spacing and baffle cut. Kern's method is used to find the heat transfer area for a given design configuration. The heat exchanger thus designed should perform the given duty subject to some pressure drop constraints and have the minimum heat transfer area.

Keywords: Heat exchanger design, Shell-and-tube heat exchanger, Double pipe heat exchanger, Optimization, Genetic algorithms.

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CHAPTER 1

INTRODUCTION

1.1 Basics of Heat Exchanger Design

The heat exchanger is an important component of any energy system. Development of design techniques for a heat exchanger with minimized cost is a vital task. Transfer of heat between two process streams is the most commonly encountered operation in process plant design. In heat exchanging equipments, heat is transferred primarily by convection from one fluid to another and the fluids are separated by a wall through which the heat is transferred. Such equipment takes many forms, of which the double pipe and the shell and tube type is the most common. Heat transfer equipments are used in essentially all process industries, and there are many different types of equipments employed for transferring heat. It is important to decide for what type of equipment is suitable for a given process.It is necessary to consider the basic process design variables and also many other factors for selection of heat transfer equipment. However, it is important to consider both process design and mechanical design while preparing the specifications for heat exchangers.

Process information includes type of fluid to be used, flow rates and amount of fluids,entrance and exit temperature, amount of vaporization or condensation, operating pressures and allowable pressure drops, fouling factors, and rate of heat transfer. Mechanical information includes size of tubes, tube layout and pitch, maximum and minimum temperatures and pressures, necessary corrosion allowances, special codes involved, recommended materials of construction.

Heat exchangers often have two different flow arrangements: parallel flows and cross flows. Incross flow arrangement, the flow lengths encountered by the two streams are independent, and can have different values. Therefore, the allowable pressure drops of the two streams can be fully utilized in the design. Shell-and-tube exchangers act like cross flow exchanger due to the baffle arrangement on the shell side. Optimal design of heat exchangers can be generally divided into two categories; design with fixed allowable pressure drops and complete optimal design. Inthe design with allowable pressure drops, the design objective is to make full utilization of theavailable pressure drops. In the complete optimal design, pressure drops are no longer set ahead of the design, and the design objective is to achieve minimum cost for the exchanger. Thus pressure drops are decided through trade-off optimization during the design process.

Designing of a Double Pipe Heat Exchanger and Shell-and-Tube Heat Exchanger (STHE) can be treated in a few subsequent phases:

1- Geometric Designing;

2- Checking;

− thermo-hydraulic calculation;

− mechanical calculation;

− techno-economic calculation;

3- Optimization;

Designing is determining the heat exchanger geometry enabling the heat exchange ratebetween hot and cold fluid, in the frame of the given operating conditions of apparatus.By checking one can investigate whether the HE of defined geometry (shell diameter,tube diameter, length of tubes, number and arrangement of tubes in bundle, number of passes for shell-side and tube-side fluid, number of baffles, ...) can perform the heat exchange between hot and cold fluid for prescribed pressure drop (bounded by allowed pressure drop) or not, i.e. is it possible to reach wanted temperature variation of fluids in given apparatus.

The aim of optimization is to adopt such a heat exchanger which could be able toperform the basic function and also be reliable in operation with satisfying economic criteria.

1.2 Optimization

The objective of optimization is to seek values for a set of parameters that maximize or minimize objective functions subject to certain constraints. Choice of values for the set of parameters that satisfy all constraints is called a *feasible solution*. Feasible solutions with objective function value(s) as good as the values of any other feasible solutions are called *optimal solutions.* Optimization techniques are used on a daily base for industrial planning, resource allocation, scheduling, decision making, etc. Furthermore, optimization techniques are widely used in many fields such as business, industry, engineering and computer science. Research in the optimization field is very active and new optimization methods are being developed regularly. Optimization encompasses both maximization and minimization problems. Any maximization problem can be converted into a minimization problem by taking the negative of the objective function, and *vice versa*. Here our problem is heat exchanger area minimization. The minimization problem can be defined as follows-

Given $f : \boldsymbol{S} \rightarrow \Re$ where $\boldsymbol{S} \subseteq \Re^{Nd}$ and Nd is the dimension of thesearch space $\boldsymbol{S}$

find $\boldsymbol{x}* \in \boldsymbol{S}$ such that $f(\boldsymbol{x}*) \leq f(\boldsymbol{x}), \forall \boldsymbol{x} \in \boldsymbol{S}$.

1.3 Optimization Algorithms

These are basically divided into two groups-

1:-Traditional methods

2:-Non-traditional methods

Traditional methods:-These are helpful in finding the optimum solution of continuous & differentiable functions. These methods are analytical & make use of the techniques of differential calculus. It provides a good understanding of the properties of the minimum & maximum points in a function & how optimization algorithms work iteratively to find the optimum point in a problem. It is classified into 2 categories-

1 .***Direct method-***

Bracketing methods, Exhaustive search method, Bounding phase method, Region-elimination method, Interval halving method, Fibonacci search method, Point estimation method, Successive quadratic method.

*2.**Gradient method***

Newton-Raphson method, Bisection method, Secant method, Cubic search method.

Nontraditional optimization algorithm:

These are quite new methods & are becoming popular day by day. Two such algorithms are-

- Genetic Algorithm

- Simulated Annealing

1.4-Objective

The objective of this project work is to design double pipe and shell and tube heat exchangers which perform a given heat duty subject to pressure drop constraints and having optimum heat transfer area.

CHAPTER 2

LITERATURE REVIEW

2.1 Double Pipe Heat Exchanger

A double pipe heat exchanger, in its simplest form is just one pipe inside another larger pipe. One fluid flows through the inside pipe and the other flows through the annulus between the two pipes. The wall of the inner pipe is the heat transfer surface. The pipes are usually doubled back multiple times as shown in the diagram at the left, in order to make the overall unit more compact.The term 'hairpin heat exchanger' is also used for a heat exchanger of the configuration in the diagram. A hairpin heat exchanger may have only one inside pipe, or it may have multiple inside tubes.The principal disadvantage to the use of double pipe exchangers lies in the small amount of heat-transfer surface contained in a single hairpin. The time and expense required for dismantling and periodically cleaning are prohibitive compared with other types of equipment. However, the double pipe exchanger is of greatest use where the total required heat-transfer surface is small, 100 to 200 ft^2 or less.[3]

Types of Double Pipe Heat Exchangers **:-**

1. Counter flow
2. Parallel Flow Heat Exchanger

1. Counter flow:-

The main advantage of a hairpin or double pipe heat exchanger is that it can be operated in a true counter flow pattern, To get More Efficiency, In the mean Time, it will give the highest overall heat transfer coefficient for the double pipe heat exchanger design.

2. Parallel Flow:-

Parallel Flow double pipe heat exchangers are focused to handle high pressures and temperatures applications. Also we can Achieve High Log mean Temperature using this.

Figure 2.1-Double Pipe Heat Exchanger

2.1.1Film Coefficients for Fluids in Pipes and Tubes

Sieder and Tate made a correlation of both heating and cooling a number of fluids, principally petroleum fractions, in horizontal and vertical tubes and arrived at an equation for streamline flow where *DG/μ*< 2100 in the form of-

$$\frac{h_i D}{k} = 1.86\left[\left(\frac{DG}{\mu}\right)\left(\frac{c\mu}{k}\right)\left(\frac{D}{L}\right)\right]^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} = 1.86\left(\frac{4}{\pi}\frac{wc}{kL}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

where, h_i=heat transfer coefficient.

D=diameter of pipe.

k=thermal conductivity.

G=mass velocity.

c=specific heat capacity.

μ=viscosity.

L=total length of the heat-transfer path before mixing occurs.

w=mass flow rate.

The above equation gives maximum mean deviations of approximately ± 12per cent from Re = 100 to Re = 2100 except for water. Beyond the transition range, the data may be extended to turbulent flow in the form of-

$$\frac{h_i D}{k} = 0.027 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

The above equations give maximum mean deviations of approximately+15and -10 per cent for the Reynolds numbers above 10,000. While these equations were obtained for tubes, they will also be used indiscriminately for pipes. Pipes are rougher than tubes and produce more turbulence for equal Reynolds numbers. Coefficients calculated from tube-data correlations are actually lower and safer than corresponding calculations based on pipe data and there are no pipe correlations in the literature soextensive as tube correlations. Equations are applicable for organic liquids, aqueous solutions, and gases.[3]

2.1.2 Fluids Flowing in Annuli: The Equivalent Diameter

When a fluid flows in a conduit having other than a circular cross section, such as an annulus, it is convenient to express heat-transfer coefficients and frictionfactors by the same types of equations and curves used for pipes and tubes. To permit this type of representation for annulus heat transfer,it has been found advantageous to employ an equivalentdiameter D_e.The equivalent diameter is four times the hydraulic radius, and the hydraulic radius is, in turn, the radius of a pipe equivalent to the annuluscross section. The hydraulic radius is obtained as the ratio of the flowarea to the wetted perimeter.[3]

2.1.3 Film Coefficients for Fluids in Annuli

The equivalent diameter is substituted in place of D in the equations for determining the heat transfer coefficient of tubes and pipes. Even though D differs from D_e, $h_o$is effective at the outside diameter of the inner pipe. In double pipe exchangers it is customary to use the outside surface of the inner pipe as the reference surface in Q = UAΔt,and since $h_i$has been determined for $A_i$and not A, it must be corrected. $h_i$is based on the area corresponding to the inside diameter where the surface per foot of length is π X ID. On the outside of the pipe the surface per foot of length is πX OD; and again letting h_{io}be the value of $h_i$referred to the outside diameter ,

$$h_{io} = h_i \frac{A_i}{A} = h_i \frac{\mathrm{ID}}{\mathrm{OD}}$$

2.1.4 Pressure Drop in Pipes and Pipe Annuli

The pressure-drop allowance in an exchanger is the static fluid pressure which may be expended to drive the fluid through the exchanger. The pump selected for the circulation of a process fluid is one which develops sufficient head at the desired capacity to overcome the frictional losses caused by connecting piping, fittings, control regulators, and the pressure drop in the exchanger itself. To this head must be added the static pressure at the end of the line such as the elevation or pressure of the final receiving vessel. Once a definite pressure drop allowance has been designated for an exchanger as apart of a pumping circuit, it should always be utilized as completely as possible in the exchanger, since it will otherwise be blown off or expanded through a pressure reducer. It is customary to allow a pressure drop of 5 to 10 psi for an exchanger or battery of exchangers fulfilling a single process service except where the flow is by gravity. For each pumped stream 10 psi is fairly standard.[6]

The pressure drop in pipes can be computed from the Fanning equation using an appropriate value of *f*. For the pressure drop in fluids flowing in annuli, replace D in the Reynolds number by $D_e$to obtain *f*. The Fanning equation may then be modified to give-

$$\Delta F = \frac{4fG^2L}{2g\rho^2D'_e}$$ [3].

2.1.5 The Calculation of a Double Pipe Exchanger

Process conditions required:

Hot fluid: T_1,T_2,*W* ,C , s or ρ, μ,k,ΔP;

Cold fluid: t_1, t_2, *W*, C, s or ρ, μ, *k*, ΔP;

The diameter of the pipes must be given or assumed.

<u>A convenient order of calculation is</u>:

1- Check the heat balance from - $Q = WC(T_1 - T_2) = wc(t_2 - t_1)$. Radiation losses from the exchanger are usually insignificant comparedwith the heat load transferred in the exchanger.

2- Calculate LMTD.

3- Calculate h_i and h_{io} from equations given above.

4- Calculate overall heat transfer coefficient U from h_i and $h_{io.}$

5- Calculation of ΔP- ΔP can be found by using the Fanning equation.

2.2 Shell and Tube Heat Exchanger

It is essential for the designer to have agood working knowledge of the mechanicalfeatures of STHEs and how they influencethermal design. The principal components of an STHE are:

1- Shell- Shell diameters are standardised.For shells including 23in. the diameters are fixed in accordance with American Society of Testing and Materials(ASTM) pipe standards. Standard inside diameters are 8,10,12,13.25,15.25,17.25,18,19.25,21.25,and 23.25 in., then 25,27 in. and so on in 2-in. increments[5]. These shells are constructed of rolled plates.

2-Tubes and tube sheets- Tubes are drawn to definite wall thickness in terms of Birmingham Wire Gauge (BWG) and true outside diameter (OD), and they are available in all common metals. Standard lengths of tubes for heat exchanger construction are 8,12,16 and 20 ft. Tubes are arranged in a triangular or square layout, known as *triangular pitch* or *square pitch* (pitch is the distance between centers of adjacent tubes). TEMA standards specify a minimum pitch of 1.25 times the outside diameter of the tubes for triangular pitch and a minimum cleaning lane of 0.25 inches for square pitch[5].

Tube OD, in.	BWG	Wall thick-ness, in.	ID, in.	Flow area per tube, in.²	Surface per lin ft, ft²		Weight per lin ft, lb steel
					Outside	Inside	
½	12	0.109	0.282	0.0625	0.1309	0.0748	0.493
	14	0.083	0.334	0.0876		0.0874	0.403
	16	0.065	0.370	0.1076		0.0969	0.329
	18	0.049	0.402	0.127		0.1052	0.258
	20	0.035	0.430	0.145		0.1125	0.190
¾	10	0.134	0.482	0.182	0.1963	0.1263	0.965
	11	0.120	0.510	0.204		0.1335	0.884
	12	0.109	0.532	0.223		0.1393	0.817
	13	0.095	0.560	0.247		0.1466	0.727
	14	0.083	0.584	0.268		0.1529	0.647
	15	0.072	0.606	0.289		0.1587	0.571
	16	0.065	0.620	0.302		0.1623	0.520
	17	0.058	0.634	0.314		0.1660	0.469
	18	0.049	0.652	0.334		0.1707	0.401
1	8	0.165	0.670	0.355	0.2618	0.1754	1.61
	9	0.148	0.704	0.389		0.1843	1.47
	10	0.134	0.732	0.421		0.1916	1.36
	11	0.120	0.760	0.455		0.1990	1.23
	12	0.109	0.782	0.479		0.2048	1.14
	13	0.095	0.810	0.515		0.2121	1.00
	14	0.083	0.834	0.546		0.2183	0.890
	15	0.072	0.856	0.576		0.2241	0.781
	16	0.065	0.870	0.594		0.2277	0.710
	17	0.058	0.884	0.613		0.2314	0.639
	18	0.049	0.902	0.639		0.2361	0.545
1¼	8	0.165	0.920	0.665	0.3271	0.2409	2.09
	9	0.148	0.954	0.714		0.2498	1.91
	10	0.134	0.982	0.757		0.2572	1.75
	11	0.120	1.01	0.800		0.2644	1.58
	12	0.109	1.03	0.836		0.2701	1.45
	13	0.095	1.06	0.884		0.2775	1.28
	14	0.083	1.08	0.923		0.2839	1.13
	15	0.072	1.11	0.960		0.2896	0.991
	16	0.065	1.12	0.985		0.2932	0.900
	17	0.058	1.13	1.01		0.2969	0.808
	18	0.049	1.15	1.04		0.3015	0.688
1½	8	0.165	1.17	1.075	0.3925	0.3063	2.57
	9	0.148	1.20	1.14		0.3152	2.34
	10	0.134	1.23	1.19		0.3225	2.14
	11	0.120	1.26	1.25		0.3299	1.98
	12	0.109	1.28	1.29		0.3356	1.77
	13	0.095	1.31	1.35		0.3430	1.56
	14	0.083	1.33	1.40		0.3492	1.37
	15	0.072	1.36	1.44		0.3555	1.20
	16	0.065	1.37	1.47		0.3587	1.09
	17	0.058	1.38	1.50		0.3623	0.978
	18	0.049	1.40	1.54		0.3670	0.831

Table 2.1 Heat Exchanger and Condenser Tube Data

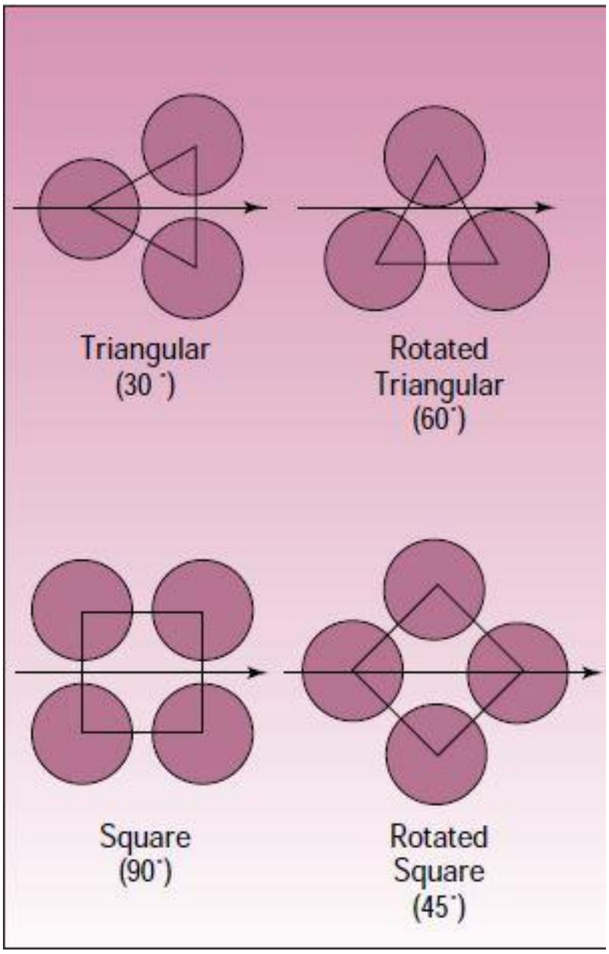


Figure 2.2 Tube Layout pattern

3-Baffle-Baffles are used to support tubes, enable a desirable velocity to be maintained for the shellsidefluid, and prevent failure of tubes due to flow-induced vibration. There are two types of baffles: plate and rod.

Baffle Spacing- Baffle spacing is the centerline-to-centerline distance between adjacent baffles. It is the most vital parameter in STHE design. The TEMA standards specify theminimum baffle spacing as one-fifth of the shell inside diameter or 2 in., whichever is greater[7]. Closer spacing will result in poor bundle penetration by the shellside fluid and difficulty in mechanically cleaning the outsides of the tubes. Furthermore, a low baffle spacing results in a poor stream distribution.

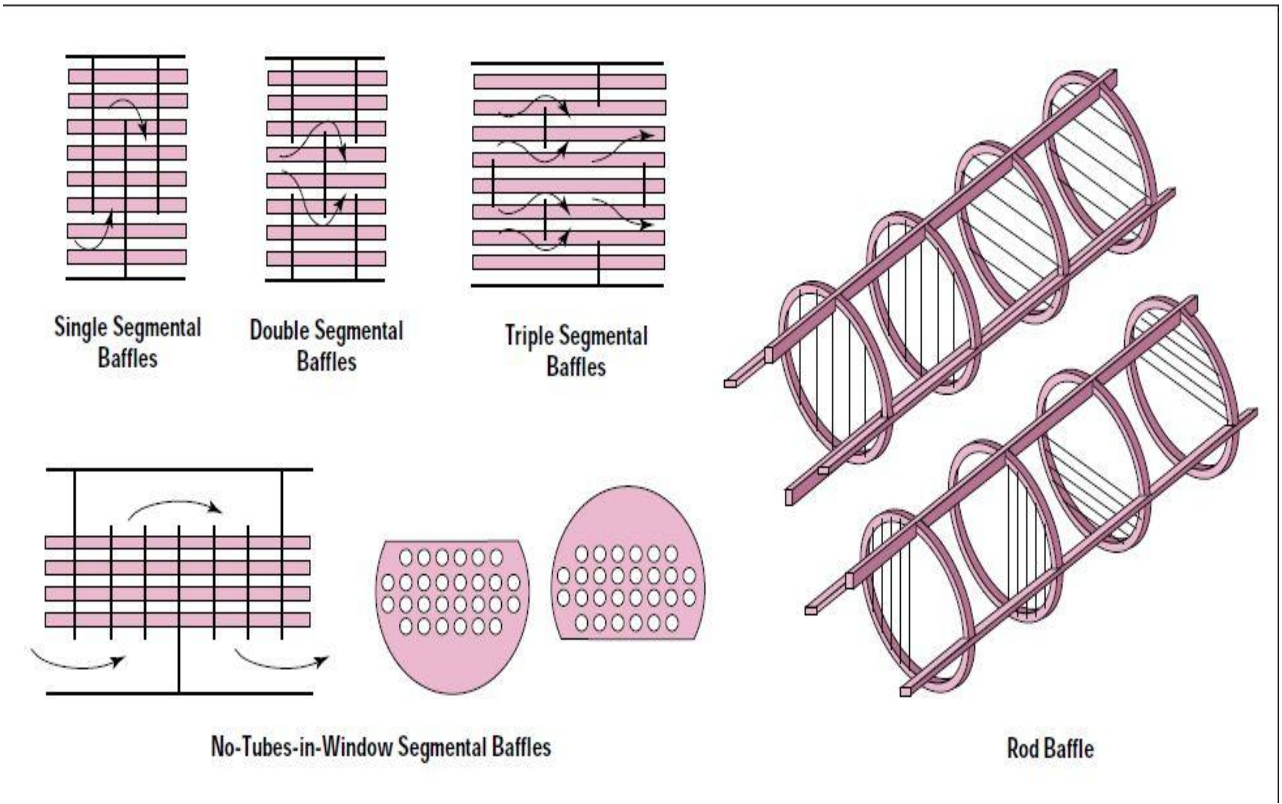


Figure 2.3- Types of Baffles

4- Channel

5- Channel cover

6- Nozzle

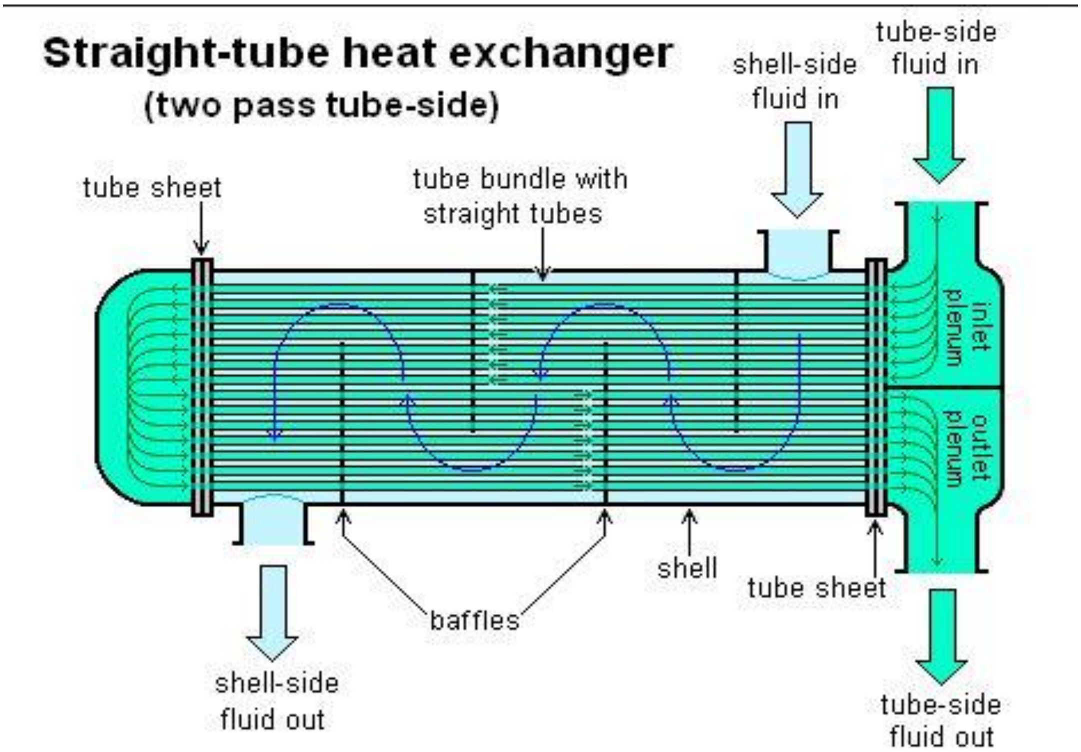


Figure 2.4- Shell and tube heat exchanger

2.2.1 THE CALCULATION OF SHELL-AND-TUBE EXCHANGERS

1- Shell-side Film Coefficients

The heat-transfer coefficients outsidetube bundles are referred to as shell-side coefficients. When the tube bundle employs baffles directing the shell-side fluid across the tubes from top to bottom or side to side, the heat-transfer coefficient is higher than for undisturbed flow along the axes of the tubes. The higher transfer coefficients result from the increased turbulence[2]. In addition to the effects of the baffle spacing the shell-side coefficient is also affected by the type of pitch, tube size, clearance, and fluid-flow characteristics. For values of Re from 2000 to 1,000,000 the data are closely represented by the equation-

$$\frac{h_o D_e}{k} = 0.36 \left(\frac{D_e G_s}{\mu}\right)^{0.55} \left(\frac{c\mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

whereh_o, $D_e$and $G_s$are as defined below.

A- Shell-side MassVelocity- The shell-side or bundle crossflow area *a*, is given by-

$$a_s = \frac{\mathrm{ID} \times C'B}{P_T \times 144} \quad \mathrm{ft}^2$$

Where, ID= inside diameter of tube (feet).

C'=clearance.

B=baffle space.

P_T=tube pitch.

Mass velocity is given as-

G_s=W/a_s

Where, W= mass flow rate of fluid.

B- Shell-side Equivalent Diameter- The equivalent diameter for the shell is then taken as four times the hydraulic radius obtained for the pattern as layed out on the tube sheet.

For square pitch-

$$d_e = \frac{4 \times (P_T^2 - \pi d_0^2/4)}{\pi d_0} \quad \mathrm{in.}$$

For triangular pitch-

$$d_e = \frac{4 \times (\frac{1}{2}P_T \times 0.86P_T - \frac{1}{2}\pi d_0^2/4)}{\frac{1}{2}\pi d_0} \quad \mathrm{in.}$$

2-True Temperature Difference-

$$\Delta T_m = F_t \Delta T_{lm}$$

Where, ΔT_{lm} is LMTD.

F_t=correction factor.

$$F_T = \frac{\sqrt{R^2+1}\,\ln\,(1-S)/(1-RS)}{(R-1)\,\ln\frac{2-S(R+1-\sqrt{R^2+1})}{2-S(R+1+\sqrt{R^2+1})}}$$

$$R = \frac{(T_1-T_2)}{(t_2-t_1)} \qquad S = \frac{(t_2-t_1)}{(T_1-t_1)}$$

where, t_1,T_1=entrance temperature of cold stream and hot stream respectively

t_2,T_2= exit temperatures of cold stream and hot stream respectively.

2- Shell-side Pressure Drop

The isothermal equation for the pressure drop of a fluid being heated or cooled and including entrance and exit losses is-

$$\Delta P_s = \frac{fG_s^2 D_s(N+1)}{2g\rho D_e \phi_s} = \frac{fG_s^2 D_s(N+1)}{5.22\times 10^{10} D_e s\phi_s} \quad \text{psf}$$

where, f=fanning friction factor

G_s= mass velocity

D_e=equivalent diameter

N+1=number of crosses

g=acceleration due to gravity

ρ=density of fluid

s=specific gravity of fluid

3-Tube-side Pressure Drop-

The pressure drop in tubes is given by-

$$\Delta P_t = \frac{fG_t^2 Ln}{5.22\times 10^{10} D_e s\phi_t} \quad \text{psf}$$

The change of direction introduces an additional pressure drop ΔP_r,called the return loss and accounted for by allowing four velocity heads per pass.

$$\Delta P_r = \frac{4n}{s}\frac{V^2}{2g'} \quad \text{psi}$$

where,V=velocity,fps.

s= specific gravity.

The total tube side pressure loss is-

$\Delta P=\Delta P_t+\Delta P_r$.

2.3 Optimization

Almost any problem in the design, operation, and analysis of manufacturingplants, and any associated problem can be reduced in the final analysis to the problem of determining the largest and smallest value of a function[8]. So, optimization is the act of obtaining the best result under given circumstances. In most engineering design activities the design objective could be simply to minimize cost of production or to maximize the efficiency of production. It is almost impossible to apply a single formulation procedure for all engineering design problems. Since the objective in a design problem and the associated design parameters vary from product to product, different techniques need to be used in different problems. For the reason, it is required to create a mathematical model of the optimal design problem, which then can be solved using an optimization algorithm. The steps involved are-

1- Need for optimization

2- Choose design variables- A design problem usually involves many design parameters, of which some are highly sensitive to proper working of the design. These are called *design or decisionvariables.*

3- Formulate constraints- The constraints represent some functional relationships among the design variables & other design parameters satisfying certain physical phenomenon & certain resource limitations. Constraints that represent limitations on the behavior or performance of the systems are termed *behavior or functional constraints.* Constraints that represent physical limitationson design variables such as availability, fabricability, & transportability are known as *geometric or side constraints.*

4- Formulate objective function- The criteria with respect to which the design is optimized, when expressed as a function of the design variables, is known as *criterion or merit or objective function.*

5- Set up variable bounds- There should be some minimum & maximum bounds on each design variables. It is required to confine the search algorithm within these bounds.

6- Choose an optimization algorithm

7- Obtain solution

2.4 Genetic Algorithm

Genetic Algorithm (GA) works on the theory of Darwin's theory of evolution andthe survival-of-the fittest [1]. Genetic algorithms guide the search through the solution space by using natural selection and genetic operators, such as crossover, mutation and the selection.Professor John Holland of the University ofMichigan envisaged the concept of these algorithms in the mid sixties.

GA encodes the decision variables or input parameters of the problem intosolution strings of a finite length. While traditional optimization techniques work directly with the decision variables or input parameters, genetic algorithms usually work with the coding. Genetic algorithms start to search from a population of encoded solutions instead of from a single point in the solution space. The initial population of individuals is created at random. Genetic algorithms use genetic operators to create Global optimum solutions based on the solutions in the current population. The most popular genetic operators are (1) selection, (2) crossover and (3) mutation. The newly generated individuals replace the old population, and the evolution process proceeds until certain termination criteria are satisfied.

2.4.1 Selection

The selection procedure implements the natural selection or the survival-of-the fittestprinciple and selects good individuals out of the current population for generating the next population according to the assigned fitness. The existing selection operators can be broadly classified into two classes: (1) proportionate schemes, such as roulette-wheel selection and stochastic universal selection and (2) ordinal schemes, such as tournament selection and truncation selection. Ordinal schemes have grown more and more popular over the recent years, and one of the most popular ordinal selection operators is tournament selection. After selection, crossover and mutation recombine and alter parts of the individuals to generate new solutions.

2.4.2 Crossover

Crossover, also called the recombination operator, exchanges parts of solutionsfrom two or more individuals, called parents, and combines these parts to generate new individuals, called children, with a crossover probability. There are a lot of ways to implement a recombination operator. The well-known crossover operators include one-point crossover. When using one-point crossover, only one crossover point is chosen at random, for example let there be two parent string A1 and A2 as:

A_1= 1 1 1 1| 1 1

A_2 = 0 0 0 0|0 0

Then, one-point crossover recombines A_1 and A_2 and yields two offsprings A_{-1} and A_{-2}as:

A_{-1} =1 1 1 1|1 1
A_{-2} =0 0 0 0|1 1

2.4.3 Mutation

Mutation usually alters some pieces of individuals to form perturbed solutions. Incontrast to crossover, which operates on two or more individuals, mutation operates on a single individual. One of the most popular mutation operators is the bitwise mutation, in which each bit in a binary string is complemented with a mutation probability.

2.4.4 Step-by-Step Implementation of GA[8]

Step 1: Initialize GA parameters which are necessary for the algorithm. Theseparameters include population size which indicates the number of individuals, number of generations necessary for the termination criterion, crossover probability, mutation probability, number of design variables and respective ranges for the design variables. If binary version of GA is used then string length is also required as the algorithm parameter.

Step 2: Generate random population equal to the population size specified. Each population member contains the value of all the design variables. This value of design variable is randomly generated in between the design variable range specified. In GA, population means the group of individuals which represents the set of solutions.

Step 3: Obtain the values of the objective function for all the population members. The value of the objective function so obtained indicates the fitness of the individuals. If the problem is a constrained optimization problem then a specific approach such as static penalty, dynamic penalty and adaptive penalty is used to convert the constrained optimization problem into the unconstrained optimization problem.

Step 4: This step is for the selection procedure to form a mating pool which consists of the population made up of best individuals. The commonly used selection schemes are roulette-wheel selection, tournament selection, stochastic selection, etc. The simplest and the commonly used selection scheme is the roulette-wheel selection, where an individual is selected for the mating pool with the probability proportional to its fitness value. The individual (solution) having better fitness value will have more number of copies in the mating pool and so the chances of mating increases for the more fit individuals than the less fit ones. This step justifies the procedure for the survival of the fittest.

Step 5: This step is for the crossover where two individuals, known as parents, are selected randomly from the mating pool to generate two new solutions known as off-springs. The individuals from the population can go for the crossover step depending upon the crossover probability. If the crossover probability is more, then more individuals get chance to go for the crossover procedure. The simplest crossover operator is the single point crossover in which a crossover site isdetermined randomly from where the exchange of bits takes place.

Step 6: After crossover, mutation step is performed on the individuals of population depending on the mutation probability. The mutation probability is generally kept low so that it does not make the algorithm unstable.

Step 7: Best obtained results are saved using elitism. All elite members are not modified using crossover and mutation operators but can be replaced if better solutions are obtained in any iteration.

Step 8: Repeat the steps (from step 3) until the specified number of generations or termination criterion is reached.

CHAPTER 3

OBJECTIVE FUNCTION FORMULATION FOR DOUBLE PIPE HEAT EXCHANGER

3.1 Double Pipe Exchanger Area Minimization

Q- It is desired to heat 9820lb/hour of cold benzene from 80^{o}F to 120^{o}F using hot toluene which is cooled from 160^{o}F to 100^{o}F.The specific gravities are 0.88 and 0.87 respectively. The allowable pressure drop on each stream is 10psi. Design a double pipe heat exchanger for this purpose while optimizing heat transfer area (and thus optimizing the heat exchanger cost) and the pressure drops on each stream. The length of the exchanger is 10 feet and the diameter should not be more than 3 feet.

Ans. – Given data-

Benzene-

ρ =0.88*62.5=55lb/ft^3

μ=1.21lb/(ft)(hr)

c=0.425Btu/lb-^{o}F

k=0.091Btu/(hr)(ft^2)(^{0}F/ft)

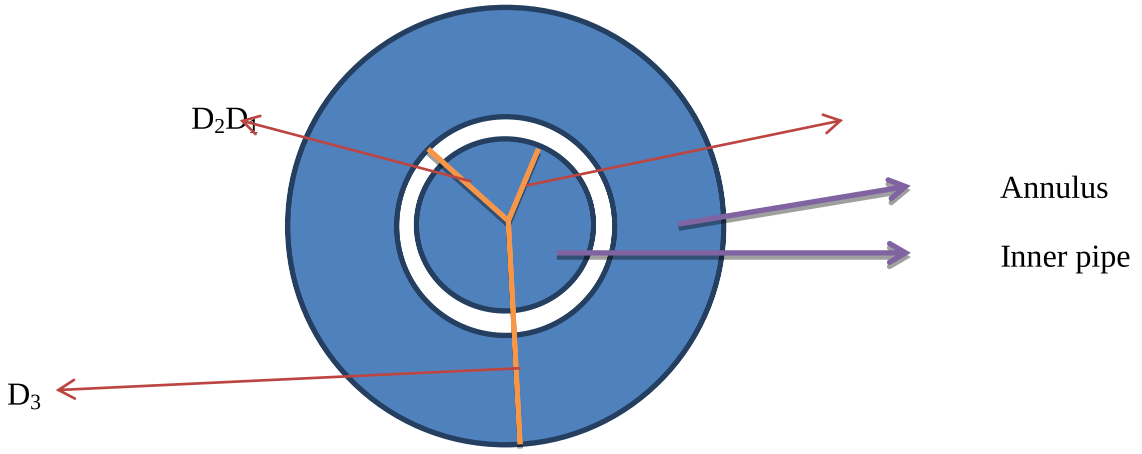
Toluene-

ρ =54.3 lb/ft^3

μ=0.99lb/(ft)(hr)

c=0.44Btu/lb-^{o}F

k=0.085Btu/(hr)(ft^2)(^{0}F/ft)



Q=9820*0.425*(120-80)=167000Btu/hr.

Hence, flow rate of toluene=167000/(0.44*(160-100))=6330lb/hr.

LMTD=20/ln(40/20)=28.8^{o}F

Cold fluid: inner pipe; benzene-

Flow area=$\pi D_1^2/4$

Mass velocity, G=(9820*4)/ πD_1^2lb/hr ft^2=12509.55/ D_1^2

Re=DG/µ

=D_1*(12509.55/ D_1^2*1.21)

=10338.47/ D_1

Pr=c µ/k=(0.425*1.21)/0.091

=5.65

Using Seider-Tate Equation-

hd/k=0.027*$Re^{0.8}$*$Pr^{.33}$

h_i=0.027*(10338.47/D_1)$^{0.8}$x1.78*(0.091/D_1)

=7.12/$D_1^{1.8}$

h_{io}=(7.12/$D_1^{1.8}$)*(D_1/D_2)

For annulus: hot fluid; toluene-

Flow area=$\pi(D_3^2$-$D_2^2)/4$

Equivalent diameter, $D_e=(D_3^2-D_2^2)/D_2$

Mass velocity, $G=(6330*4)/ \pi(D_3^2-D_2^{2)}$

$Re=DG/\mu$

$=((D_3^2-D_2^2)/D_2)*(8063.70/ D_3^2-D_2^2)*(1/0.99)$

$=8145.15/ D_2$

$Pr = c\ \mu/k=5.12$

$h_o=0.027*Re^{0.8}*Pr^{1/3}*(k/D_e)$

$=(5.29D_2^{0.2})/(D_3^2-D_2^2)$

Clean overall coefficient= $h_{io}\ h_o/(h_{io}+h_o)$

$=(37.66D_2^{0.2})/(7.12D_3^2-7.12D_2^2+5.29*D_1^{0.8}*D_2^{1.2})$

Now, $Q=UA\Delta T_L$

Hence, $A=Q/U\Delta T_L$

$=154(7.12D_3^2-7.12D_2^2+5.29*D_1^{0.8}*D_2^{1.2})/ D_2^{0.2}$

Pressure drop calculations-

$\Delta P=2fL\rho v^2/D$

Inner pipe-

$v=G/3600\rho$

$=0.063/D_1^2$

$f=0.0014+(0.125/Re^{0.32})\rightarrow$Drew, Koo and McAdams equation

$\Delta P=4.4(0.0014+0.0064*D_1^{0.32})/D_1^5$

Annulus-

$v=G/3600\rho$

$=0.041/(D_3^2-D_2^2)$

$f=0.0014+(0.125/Re^{0.32})\rightarrow$Drew, Koo and McAdams equation

D_e' for pressure drop differs from D_e for heat transfer-

$D_e'=D_3-D_2$

$Re'= D_e'G/\mu$

$=8145.15/(D_3+D_2)$

$\Delta P=(1.85*(0.0014+0.007(D_3+D_2)^{0.32}))/(D_3+D_2)^2(D_3-D_2)^2$

Now our problem can be formulated mathematically as-

Minimise $A=154(7.12D_3^2-7.12D_2^2+5.29*D_1^{0.8}*D_2^{1.2})/D_2^{0.2}$

Subject to- $4.4(0.0014+0.0064*D_1^{0.32})/D_1^5*144\leq10$

$(1.85*(0.0014+0.007(D_3+D_2)^{0.32}))/144*(D_3+D_2)^2(D_3-D_2)^2\leq10$

$D_2-D_1=0.0334$ (assuming 1 cm thick inner pipe)

$D_1, D_2, D_3\leq3$

$D_2<D_3$

$D_1, D_2, D_3>0.1$ (assuming pipe diameters to be at least 0.1 feet)

CHAPTER 4

OBJECTIVE FUNCTION FORMULATION FOR SHELL & TUBE HEAT EXCHANGER

4.1 Shell and Tube Heat Exchanger Area Minimization

Problem-43800 lb/hr of a 42^{o}API kerosene leaves the bottom of a distilling columnat 390^{o}F and will be cooled to 200^{o}F by 149000lb/hr of 34^{o}API mid-continent crude coming from storage at 100^{o}F and heated to 170^{o}F.A 10 psi pressure drop is permissible on both streams. Design a shell and tube heat exchanger having 1 in. OD, 13 BWG tubes, 16 feet long and laid out on 1.25 in. square pitch.

Solution- Given data-

Kerosene-

c=0.605 Btu/lb^{o}F

μ=0.97 lb/ft-hr

k=0.0765 Btu/(hr)(ft^2)(^{o}F/ft)

Crude Oil-

c=0.49 Btu/lb^{o}F

μ=8.7 lb/ft-hr

k=0.077 Btu/(hr)(ft^2)(^{o}F/ft)

Design Variables-

Shell side-

Inner diameter=D

Baffle Space=B

Passes=1

Number of baffles=N

Tube side-

Number of tubes=N_t

Length=16 feet

OD, BWG, Pitch=1 in.,13 BWG, 1.25 in. respectively.

Passes = n.

Heat balance-

Q=43800*0.605*(390-200)=510000Btu/hr

LMTD=152.5^{o}F

R=190/70=2.71

S=70/(390-100)=0.241

F_t=0.905

ΔT=0.905*152.5=138^{o}F

Cold fluid:Tubeside,crude oil-

Flow area=0.515 in.^{2}

$a_t=N_t a_t'/144n$

$=N_t*0.515/144n$

Mass velocity, $G_t=W/a_t$

$=149000*144n/N_t*0.515$

$=(149000*279.6)n/N_t$

D=0.81/12=0.0675 feet

$Re=DG_t/\mu$

$=0.0675*(149000*279.6)n/8.7N_t$

$=323227.2n/N_t$

$Pr=c\mu/k$

$=0.49*8.7/0.077$

$=55.36$

$Pr^{1/3}=3.81$

$h_iD_i/k=0.027*Re^{0.8}*Pr^{1/3}\rightarrow$Sieder-Tate Equation

$\Rightarrow h_i=0.027*(323227.2n/N_t)^{0.8}*3.81*(0.077/0.0675)$

$=2999.73*(n/N_t)^{0.8}$

$h_{io}=h_i*ID/OD$

$=2999.73*(n/N_t)^{0.8}*0.81$

$=2429.78(n/N_t)^{0.8}$

Hot fluid: shell side, kerosene-

Flow area=ID*C'*B/144P_t

=BD/5

Mass velocity, $G_s=W/a_s$

$=43800*5/BD$

$Re=D_eG_s/\mu$

$D_e=4*(P_t^2-\pi d_o^2/4)/\pi d_o$

$=4*(1.25^2-\pi/4)/\pi$

=0.99 in.

=0.0825 feet.

Re=0.0825*43800*5/0.97*BD

=18626.29/BD

Pr=cμ/k

=0.605*0.97/0.0765

=7.48

$Pr^{1/3}=1.95$

$h_oD_e/k=0.36*Re^{0.55}*Pr^{1/3}$

$\Rightarrow h_o=0.36*(18626.29/BD)^{0.55}*1.95*(0.0765/0.0825)$

$=145.25/(BD)^{0.55}$

Clean overall coefficient, $U_c=h_{io}h_o/(h_{io}+h_o)$

$$=\frac{2429.78*(n/N_t)^{0.8}*(145.25/(BD)^{0.55})}{2429.78(n/N_t)^{0.8}+145.25*(BD)^{0.55}}$$

$$=\frac{2429.78*n^{0.8}*145.25}{2429.78*n^{0.8}(BD)^{0.55}+145.25N_t^{0.8}}$$

$Q=U_cA\Delta T$

$\Rightarrow A=Q/U_c\Delta T$

$$=\frac{5100000*(2429.78n^{0.8}*(BD)^{0.55}+145.25N_t^{0.8}}{138*2429.78*n^{0.8}*145.25}$$

$=254.4(BD)^{0.55}+15.2*(N_t/n)^{0.8}$

Pressure Drop Calculation-

Tube side:

$$\Delta P_t=\frac{fG_t^2Ln}{144*5.22*10^{10}*D}\text{ psf.}$$

$$=\frac{(0.0014+0.125/Re^{0.32})(149000*279.6n/N_t)^2*l*n}{144*5.22*10^{10}*0.0675*0.83}$$

$$=\frac{(4121.28n^3*16)*(0.0014+(0.125N_t^{0.32}/(323227.2n)^{0.32})}{N_t^2}$$

$$= (5.77*16*n^3)/N_t^2+(8.89*n^{2.68}*16/N_t^{1.68})$$

$\Delta P_r=4nV^2/2sg'$

$$=4*n*(50005.88n^2/N_t^2)*(1/0.83*2*32.15)$$

$$=3747.93n^3/N_t^2$$

$\Delta P_T=\Delta P_t+\Delta P_r$

$$=(5.77*16*n^3)/N_t^2+(8.89*n^{2.68}*16/N_t^{1.68})+3747.93n^3/N_t^2$$

Shell side:

$$\Delta P_s=\frac{fG_s^2D(N+1)}{5.22*10^{10}*D_e*s}$$

$$=(0.0014+0.125/Re^{0.32})*(43800*5/BD)^2*D*(16/B)*(1/5.22*10^{10}*0.73*0.0825)$$

$$= (0.342/DB^3)+(1.31/D^{0.68}B^{2.68})$$

Now our problem can be formulated mathematically as-

Minimise $A=254.4(BD)^{0.55}+15.2*(N_t/n)^{0.8}$

Subject to: $(5.77*16*n^3)/N_t^2+(8.89*n^{2.68}*16/N_t^{1.68})+3747.93n^3/N_t^2\leq 10$

$(0.342/DB^3)+(1.31/D^{0.68}B^{2.68})\leq 10$

$0.2D\leq B\leq D$

$B, D, N_t, n\geq 0$

CHAPTER 5

RESULTS AND DISCUSSIONS

5.1 Solution of Double Pipe Exchanger-

The design problem was reduced mathematically to-

Minimise $A=154(7.12D_3^2-7.12D_2^2+5.29*D_1^{0.8}*D_2^{1.2})/ D_2^{0.2}$

Subject to- $4.4(0.0014+0.0064*D_1^{0.32})/D_1^5*144\leq10$

$(1.85*(0.0014+0.007(D_3+D_2)^{0.32}))/144*(D_3+D_2)^2(D_3-D_2)^2\leq10$

$D_2-D_1=0.0334$ (assuming 1 cm thick inner pipe)

$D_1, D_2, D_3\leq3$

$D_2<D_3$

$D_1, D_2, D_3>0.1$ (assuming pipe diameters to be at least 0.1 feet)

Such a minimisation problem subjected to certain constraints can be solved by using MATLAB functions such as ‘fmincon’ or ‘Genetic Algorithm’.

Solution Using ‘fmincon’-

We write 3 MATLAB codes- one for the objective function, one for the constraints and one to call the ‘fmincon’ function to solve this problem.

MATLAB code for objective function-(myfun.m)

```
function f = myfun(x)
f = (154/x(2)^0.2)*(7.12*x(3)^2-7.12*x(2)^2+5.29*x(1)^0.8*x(2)^1.2);
```

MATLAB code for constraints-(mycon.m)

```
function [c,ceq] = mycon(x)
a=(4.4/144*x(1)^5)*(0.0014+0.0064*x(1)^0.32) - 10;
b=((0.013*(0.0014+0.007*(x(2)+x(3))^0.32))/((x(2)+x(3))^2*(x(3)-x(2))^3))-10;
d=(x(1)-3);
e=(x(2)-3);
f=(x(3)-3);
g=(x(2)-x(3));
h=-x(1)+0.1;
i=-x(2)+0.1;
j=-x(3)+0.1;
c = [a;b;d;e;f;g;h;i;j] ;    % Compute nonlinear inequalities at x.
ceq =x(2)-x(1) - 0.0334;   % Compute nonlinear equalities at x.
```

MATLAB code to call ‘fmincon’ function-(fmincon.m)

```
close all
clear all
A=[];
b=[];
Aeq=[];
beq=[];
lb=[];
ub=[];
x0 = [10; 10; 10];     % Starting guess at the solution
[x,fval] = fmincon(@myfun,x0,A,b,Aeq,beq,lb,ub,@mycon)
```

Results using ‘fmincon’-

```
> In fmincon at 439
  In Opromfmincon at 10

Local minimum possible. Constraints satisfied.

fmincon stopped because the size of the current search direction is less than
twice the default value of the step size tolerance and constraints were
satisfied to within the default value of the constraint tolerance.

<stopping criteria details>

Active inequalities (to within options.TolCon = 1e-006):
  lower      upper     ineqlin   ineqnonlin
                                     2
                                     7

x =

    0.1000
    0.1334
    0.1772


fval =

   39.5155
```

D_1=0.1 feet =1.2 inch=0.03 meter

D_2=0.13 feet= 1.56 inch=0.039 meter

D_3=0.18 feet= 2.2 inch=0.055 meter

Area= 39.5 feet^2=3.5m^2

We used the MATLAB Optimization Toolbox to implement the Genetic Algorithm-

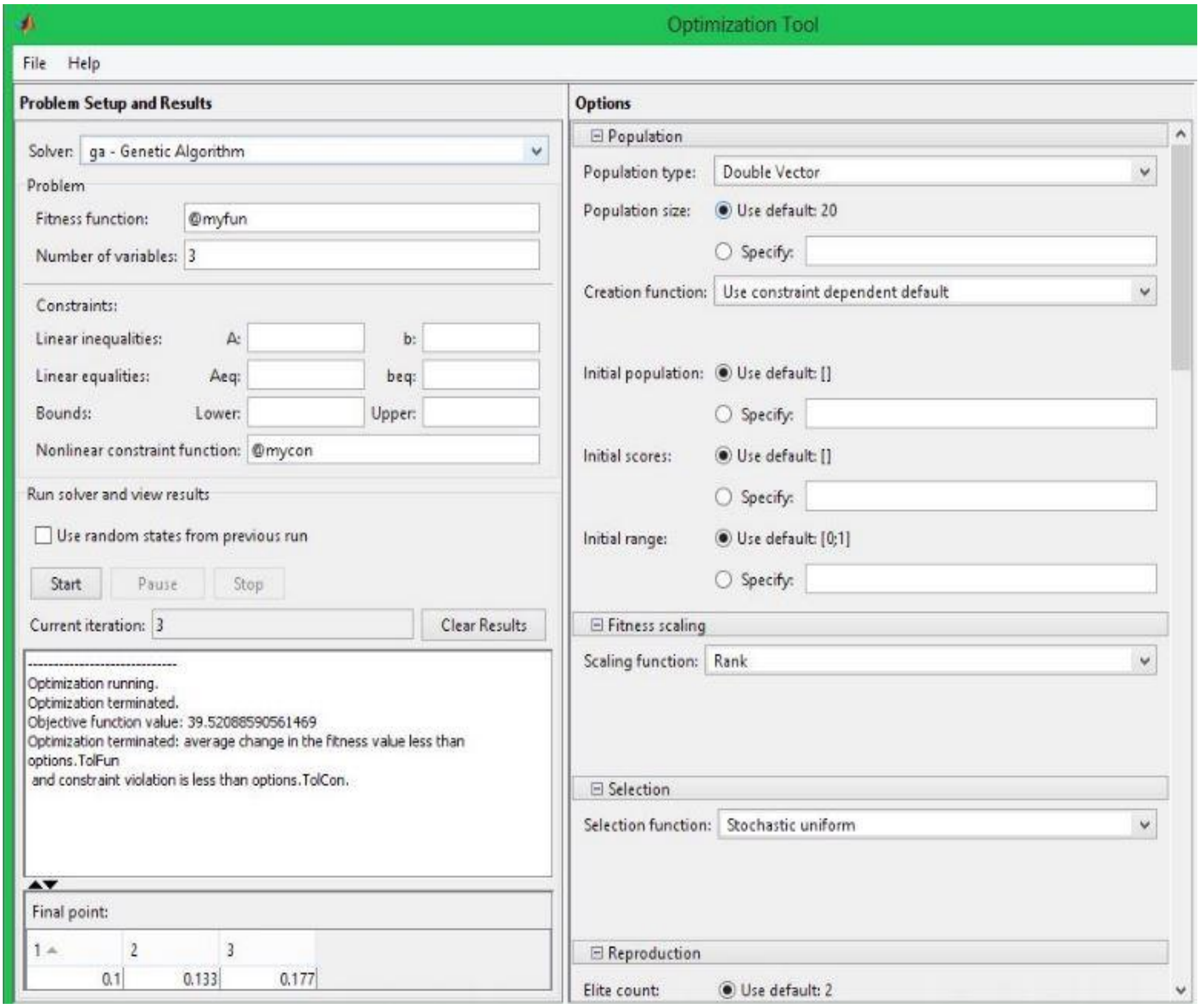


Figure 5.1 Output of the Optimization Toolbox (GA) for Double Pipe Exchanger

Results using 'Genetic Algorithm'-

D_1=0.1 feet= 1.2 inch=0.030 meter

D_2=0.133 feet= 1.6 inch=0.04 meter

D_3=0.177 feet= 2.12 inch=0.053 meter

Area= 39.52 feet^2=3.55 m^2

5.2 Solution of Shell and Tube Heat Exchanger Problem-

The design problem was reduced mathematically to-

Minimise $A=254.4(BD)^{0.55}+15.2*(N_t/n)^{0.8}$

Subject to: $(5.77*16*n^3)/N_t^2+(8.89*n^{2.68}*16/N_t^{1.68})+3747.93n^3/N_t^2 \leq 10$

$(0.342/DB^3)+(1.31/D^{0.68}B^{2.68})\leq 10$

$0.2D\leq B \leq D$

$B, D, N_t, n\geq 0$

$1\leq D \leq 4$

Such a minimisation problem subjected to certain constraints can be solved by using MATLAB functions such as ‘fmincon’ or ‘Genetic Algorithm’.

Solution Using ‘fmincon’-

We write 2 MATLAB codes- one for the objective function and another for the constraints.

MATLAB code for objective function (myfun.m)-

```
function f = myfun(x)
f = (254.4*x(1)^0.55*x(2)^0.55+15.2*x(3)^0.8*x(4)^-0.8);
```

MATLAB code for constraints-

```
function [c,ceq] = mycon(x)
a=(5.77*x(4)^3*16*x(3)^-2+8.89*x(4)^2.68*16*x(3)^-1.68+3747.93*x(4)^3*x(3)^-2)-10;
b=(0.342*x(2)^-1*x(1)^-3+1.31*x(2)^0.68*x(1)^2.68)-10;
c=0.2*x(2)-x(1);
d=-x(1);
e=-x(2);
f=-x(3);
g=-x(4);
h=1-x(2);
i=x(2)-4;
j=x(1)-x(2);
c = [a;b;c;d;e;f;g;h;i;j] ;    % Compute nonlinear inequalities at x.
ceq=[];  % Compute nonlinear equalities at x.
```

We used the MATLAB Optimization Tool box to apply ‘fmincon’ and ‘Genetic Algorithm’ to the above problem.

Result using ‘fmincon’-

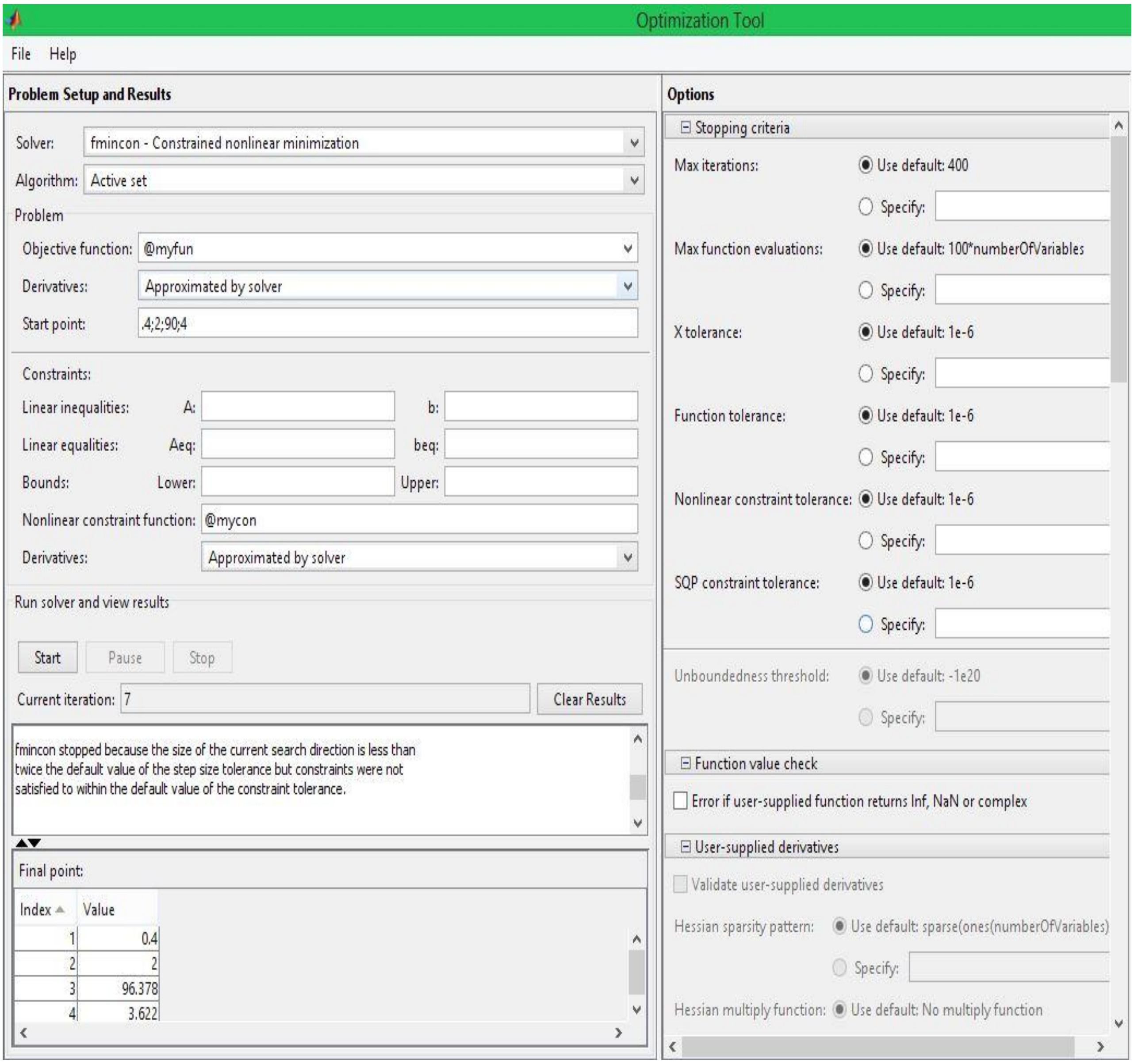


Figure 5.2 Output of the Optimization Toolbox(fmincon) for Shell & Tube Exchanger Problem

Baffle space=0.4 feet = 4.8 in.= 0.12 meter

Shell diameter=2 feet=24 in.= 0.61 meter

Number of tubes=96

Number of tube passes=4

Area=518.2 feet^2= 46.6 m^2

Result using GA-

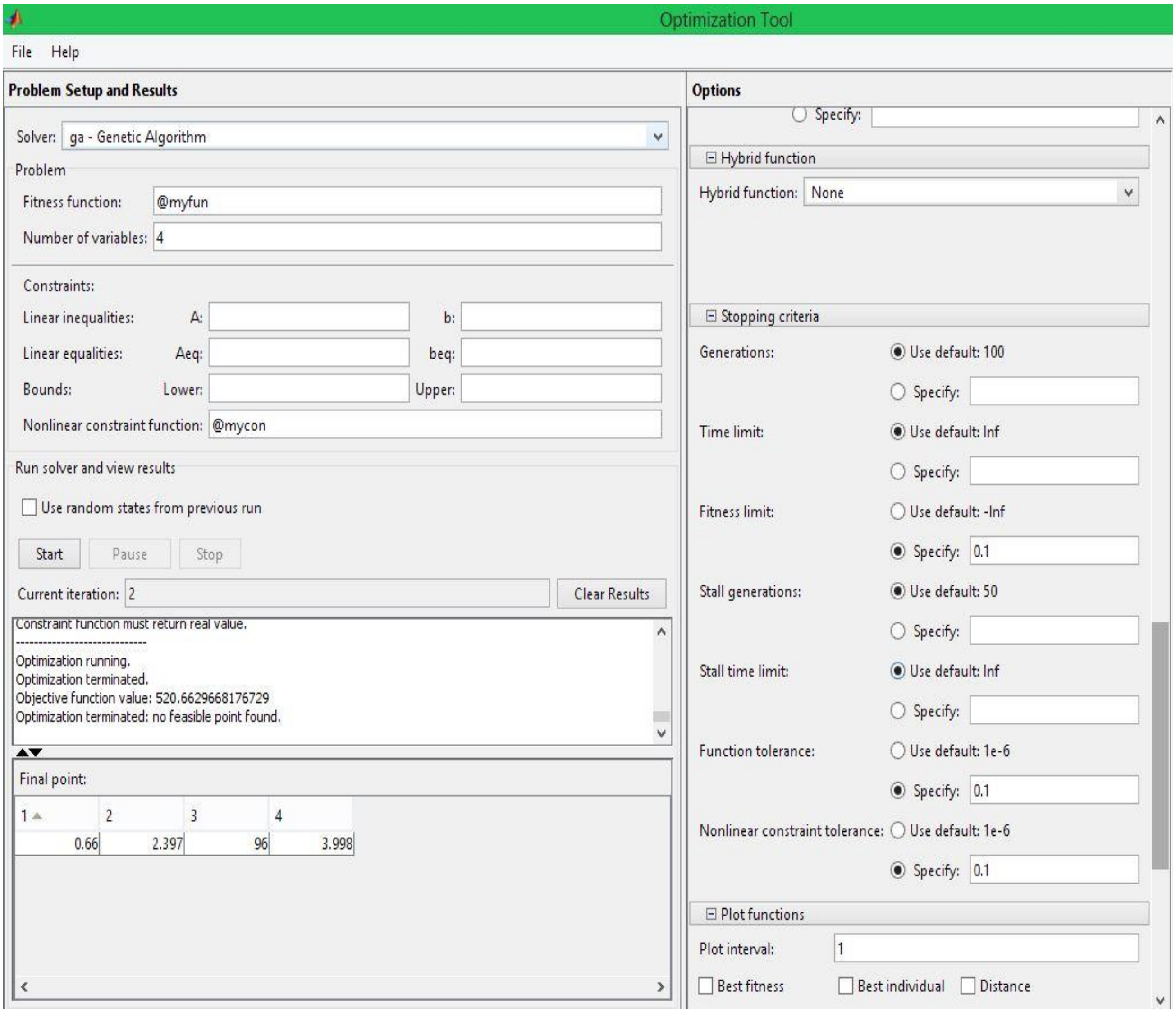


Fig. 5.3 Output of the Optimization Toolbox(GA) for Shell & Tube Exchanger

Baffle space=0.66 feet = 0.2 meter

Shell diameter=2.4 feet= 0.73 meter

Number of tubes=96

Number of tube passes=4

Area=520.66 feet^2= 46.86m^2

CHAPTER 6
CONCLUSIONS AND FUTURE WORK

6.1 CONCLUSION

This thesis work focused on application of traditional and non-traditional optimization techniques on area minimization of double pipe and shell and tube heat exchangers. A generalized procedure has been developed to run the GA algorithm coupled with a function that uses Kern's method of heat exchanger design, to find the global minimum heat exchanger area.The objective functionis the area obtained by using Kern's method and genetic algorithm optimization method is applied to solve the multivariable optimization problems which not only yields the globe optimum solution but also demonstrates the flexibility to select the design variables and constraint conditions.The design variables which areused for the optimization of shell and tube heat exchanger are shell inside diameter,number of tubes, baffle spacing and number of tube passes. The design variables which areused for the optimization of double pipe exchanger are the pipe diameters of inner and outer pipe. The optimization and analysis of these design parameters are very important for the better performance of heat exchanger.

6.2 FUTURE WORK

More elaborate methods like the Bell Delaware method of heat exchanger design could be used for objective function formulation. Also other non-traditional optimization algorithms like Particle Swarm Optimization could be used for obtaining the optimum design which are much faster and have better probability of arriving at the global optimum solution.

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